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Leakage Experiments on a Running Twin Screw Compressor

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ABSTRACT

This paper presents experiments in which the indicated diagram is measured with different sizes of the house sealing line gap. The measurements and the data analysis will be explained, after motivating the choice for this particular leakage gap and a short discussion of the test-rig. The central theme is the discussion of the influences of the leakage on the compression line of the indicated diagram. It will be shown that these experiments have the potential to quantify the leakage phenomena inside the compressor.

INTRODUCTION

Relatively little is known about leakage in screw compressors. This is surprising because leakage is generally considered to be one of the major sources of efficiency loss. One of the reasons that so little is known is the difficulty of measuring the leakage. Some work on this topic has been carried out, see eg. Peveling (1) or Sachs (2). but only on static models and disregarding liquid injection.

The approach taken in this investigation is to change only one parameter, the size of one of the leakage gaps. As a result, all differences between measured indicated diagrams can only be contributed to this gap. This makes it possible to isolate the effect of this leakage. With the use of computer simulation, this opens the door to quantification of the flow coefficients. It also gives the opportunity to verify different models for the leakage.

SELECTION OF THE LEAKAGE PATH

There are six leakage paths that are directly related to the rotor profiles. It lies not within the scope of this paper to give a full discussion of these, please refer to the general literature on screw compressors. One particularly interesting with respect to the effects of leakage is Fleming and Tang (3). The sizes of the paths cannot be changed easily for all of these paths:

The sizes of the *cusp blow hole* and the *compression start blow hole* are properties of the rotor profile. The only way to change them is by changing the design of the profile but this cannot be done without changing many other features that influence the indicated diagram and the influence of the leakage through these paths on the diagram cannot be isolated from the other effects.

The *suction and discharge end-face clearances* can easily be changed by changing the length of the house (eg. by adding a gasket) or by lathing the rotors to a shorter length. There is even some room in the compressor to move the rotors closer to the suction or the discharge end-face, increasing one gap and decreasing the other. However, simulations presented earlier by the authors (4), suggest that this hardly affects the indicated diagram.

The *contact line* between the rotors can be changed by removing some material from the rotor surfaces or by increasing the distance between the axes. This is probably the most interesting option. But this requires either high precision machining of the rotors or the production of another compressor house which could not be done within the limitations of this research.

The height of the gap from *house sealing line* is easily increased by reducing the diameter of the rotors, a relative simple operation on a lathe. This implies also a larger gap for the contact line, but this is insignificant compared to the increase in that for the house sealing line. The simulations mentioned above have shown that the effect on the indicated diagram is significant.

Considering the effects on the indicated diagram and the difficulties in realizing increases of the gap sizes, the house sealing line was selected for the experiments. The diameter of the rotors was reduced on a lathing. This results in a bigger gap between the rotor tips and the compressor housing.

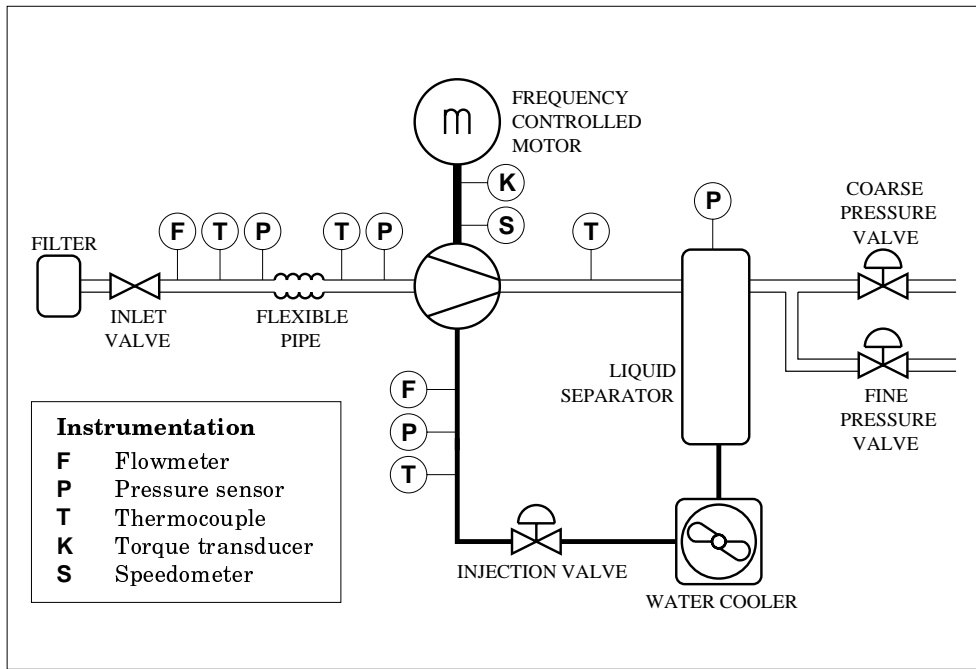


Figure 1: Schematic view of the compressed air unit.

Another leakage path appears at the labyrinth seals which are used in the compressor to relieve the pressure on the radial lip seals on the discharge side. The fluid that leaks through these labyrinths is injected in to the suction inlet of the compressor. This path is not taken into consideration here.

EXPERIMENTAL SETUP

A Grassair 105 compressor formed the heart of the test-rig. Water injection is used for cooling, lubrication (plastic rotors) and sealing. It was embedded in a compressed air unit. A frequency regulator was included to vary the speed. The instrumentation can be divided into two groups.

The first group measured the in coming and outgoing mass and energy. This part is shown in figure 1. It includes temperature and pressure sensors at the compressor inlets and outlet, torque and speed transducers on the driving axis (to measure the work), and flow meters for the injection water and the air flow. These measure the compressor performance on a relative large time scale.

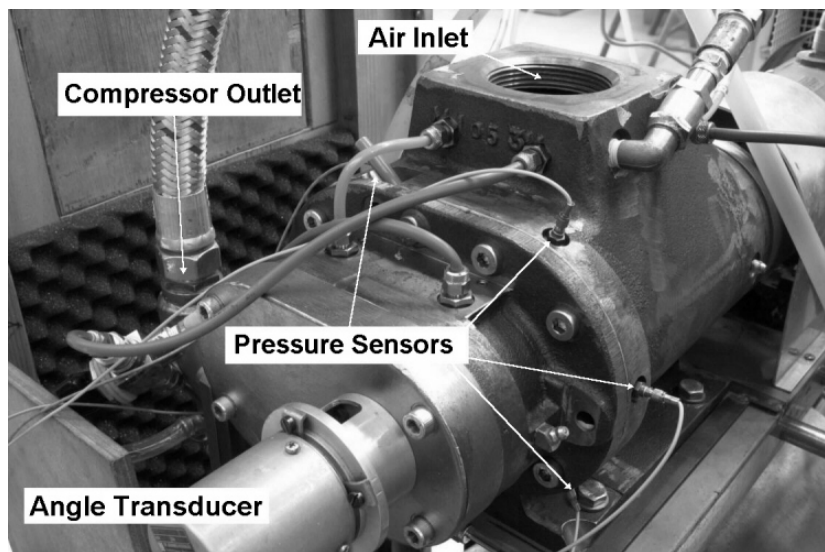


Figure 2: Grassair 105 Compressor seen from the discharge side. The positions of the angle transducer and four of the pressure sensors (the fifth being hidden underneath).

The second part of the instrumentation was laid out to measure the indicated diagram, which requires on a much shorter time scale. Or, to be more precise, to measure the compression line. During suction and discharge, the pressure is almost entirely determined by the external pressures and the flow losses through the suction and discharge ports. It is therefore not possible to isolate the effects of leakage on these parts of the diagram.

Five pressure transducers were mounted 65° apart on a circle, as shown in figure 2. Together they cover 325° of the working process, enough to record the last part of the suction, the

entire compression and the first part of discharge phase. Though the interest only goes to the compression phase, the suction and discharge pressures are useful as references. A data acquisition card in the computer was triggered 512 times per revolution to sample the five pressure sensors. This trigger was provided by an angle transducer that was mounted directly on the compressor axis. The size of the working volume was determined from the angle and analysis of the rotor profile.

DATA ANALYSIS

As indicated in the previous section, the five pressure sensors were mounted 65° apart. Because the five lobes on the male rotor are 72° apart, this gives a 7° overlap between the sensors. This feature was built in because the sensors give relative pressures: by laying the overlapping parts on top of each other, the absolute pressures would be obtained. However, oscillations following the passing of sensors by the lobes extended over a larger period of time than was foreseen. This is illustrated in figure 3. As a result, extrapolation of the useful parts of the sensor reading was unavoidable.

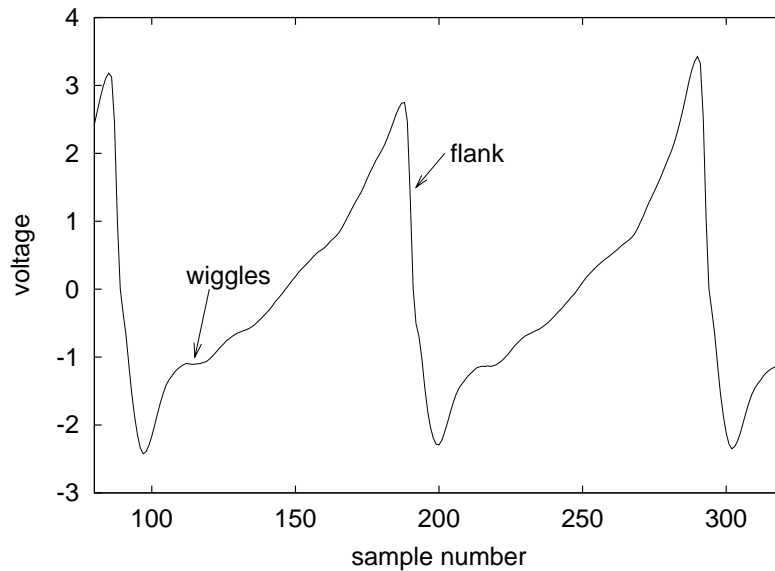


Figure 3: Reading from one of the pressure sensors. The flank denotes the passing of the rotor tip. Part of readings cannot be used because of oscillations that appear as wiggles in the signal.

This proved to be a rather troublesome task. The use of idealized compression models (such as isentropic or polytropic compression) were not possible because the interest goes to the deviations from such models. Eventually, an extrapolation method was used to breach each gap between the readings individually. It is devised such that the curves are zeroth order (no vertical steps in the diagram) and first order (no sharp angles) continuous. Figure 4 shows a typical result.

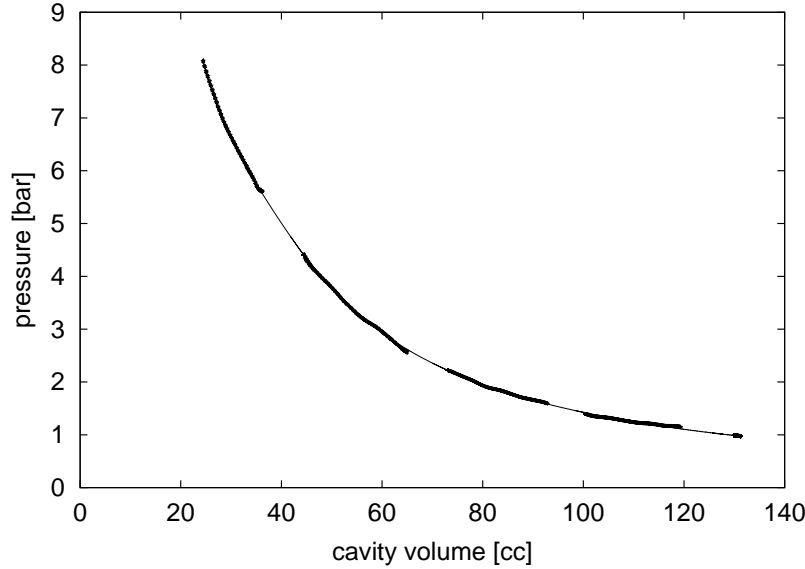


Figure 4: Compression line after removal of the wiggles and extrapolation between the signals from the different sensors.

EXPERIMENTAL RESULTS AND DISCUSSION

Some selected results will be shown in this section. They were chosen to show the feasibility of the method. The central question asked in making the selection and addressed in the discussion was: What do the measurements show of the effects of leakage?

The results that will be presented here were obtained with two different sizes of the house sealing line. After the tests with the compressor as it was originally produced, both rotors were reduced and thereby the gaps were increased with 200 micron.

Figure 5 shows the compression lines for a discharge pressure of 8 bara, which is typical in compressed air installations. The compressor was running at 4000 rpm.

Each cavity sees two sealing lines, one on the suction side through which gas escapes to the leading cavity (or the suction plenum) and one on the discharge side through which gas flow in from the trailing cavity.

At the start of the compression, the pressure difference between the suction plenum and the working cavity is small. The amount of fluid escaping from the cavity through the gap towards suction is therefore also small. The pressure difference over the house sealing line that connects to the leading pocket is much higher and the incoming flow is already large at the moment that the suction window closes. Thus more fluid is entering the cavity than escaping. The net result is a higher pressure. As these effects are stronger for the larger leakage gap, the pressure found in the early stages of compression for this large gap is higher than for the small gap.

The extra mass gathered in the beginning is lost again at a later stage. Following the same line of reasoning as above, the extra pressure is partly undone due to this loss of mass. The compression lines in the figure converge again towards the end of compression.

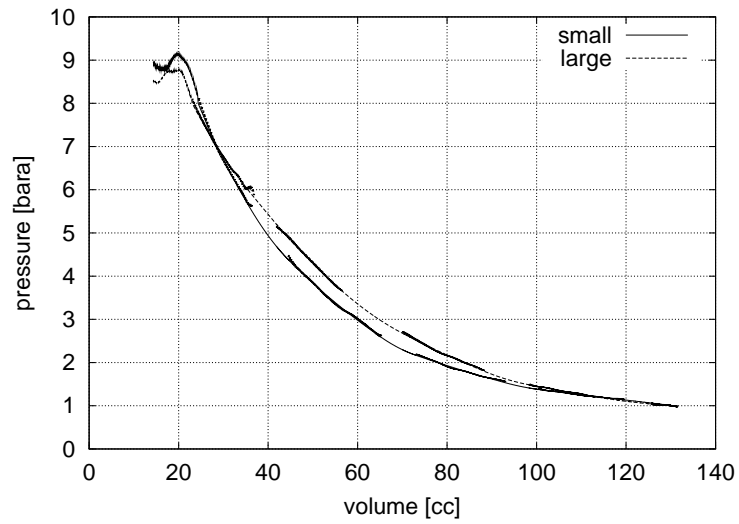


Figure 5: Indicated diagram at 8 bara, 4000 rpm with small and large house sealing line gap.

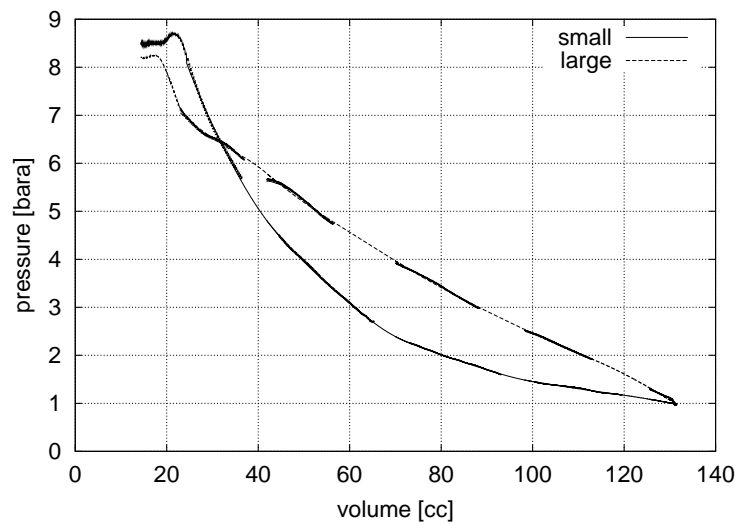


Figure 6: Indicated diagram at 8 bara, 2000 rpm with small and large house sealing line gap.

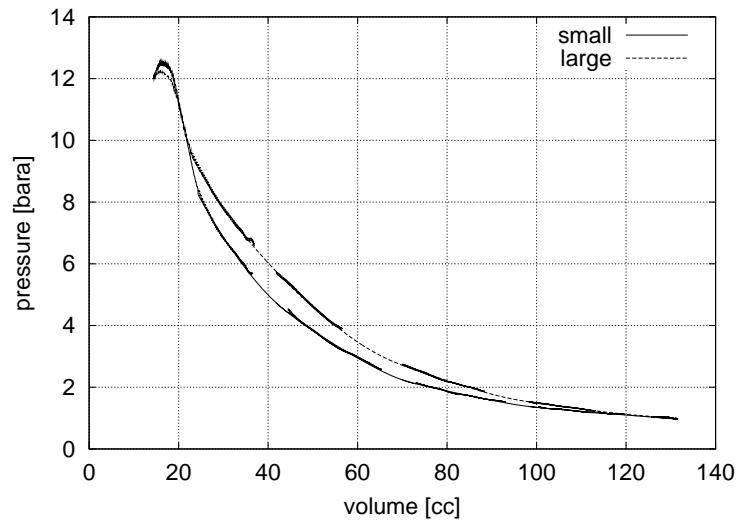


Figure 7: Indicated diagram at 11 bara, 4000 rpm with small and large house sealing line gap.

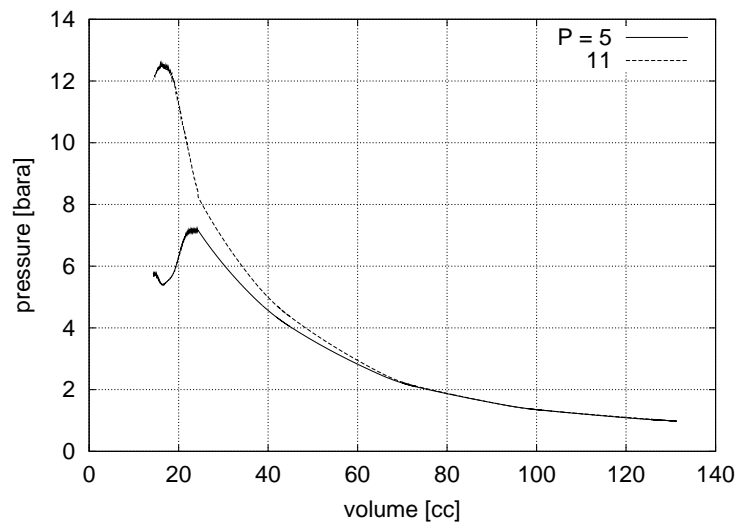


Figure 8: Indicated diagram at 5 and 11 bara, 4000 rpm both with with small house sealing line gap.

The net result is that the compression line of the large gap lies above that for the small gap. Thus more work is done. At the same time the net mass flow is smaller. Leakage back to suction still occurs and more so if the gap the larger. Thus both the isentropic and the volumetric efficiency decrease as the gap size increases.

At lower speeds, the compression process takes more time. This also means that there is more leakage. The effect of speed is very large as shown in figure 6.

The effect is also higher when the discharge pressure is higher. This is shown in figure 7. There are two reasons to be given for this. First the pressure difference between the cavities is larger and because this is the driving force of the leakage, the leakage increases. Second, the density of the fluid is higher (at least at the end of compression) which implies larger leakage mass flow. But the discharge pressure mainly affects the last part of the compression. This can be seen in figure 8, where the gaps are small and only the pressures differ.

SUMMARY AND CONCLUSIONS

An experiment was executed in which the gap size of the house sealing line of a twin screw compressor was increased. In this way, all other phenomena that play a role during compression are unchanged and the differences between the modified and the original compressor must be contributed to the leakage through this path. It has been shown that the effects of the artificially increased leakage are visible in the indicated diagram. Comparison with computer simulations must now be made in order to obtain a quantification of the leakage and a validation of the leakage model.

In this investigating the house sealing line was chosen for practical reasons. Other paths can be changed but some will either give little effect or will require such compressor modifications that isolation of the effect is no longer possible. However, there are methods to change the clearance between the rotors.

ACKNOWLEDGMENT

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